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샤시 제어를 위한 동적 목표 요레이트 설계

Dynamic Target Yaw-rate Design for Chassis Control

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기계항공공학부

박 관 우

Abstract

Dynamic Target Yaw-rate Design for Chassis Control

Kwanwoo Park
School of Mechanical and Aerospace Engineering
The Graduate School
Seoul National University

This study presents a dynamic target yaw-rate design method for chassis control system. The target yaw-rate is essential for the supervisor of the Integrated Chassis Control(ICC) algorithm.

The supervisory controller monitors the vehicle status and determines desired vehicle motions such as a target yaw-rate. The target design is important because the inputs such as lateral force, yaw moment are calculated according to this target motion in the upper and lower level controller. Conventional target design is parameter optimization for a specific scenario and road condition. However, this has the disadvantage of lacking interchangeability between different scenarios.

In this work, research has been conducted to make the target yaw-rate design universal. The proposed design method consists of two parts: A bicycle model, which is considered transient handling characteristic, and Relaxation Length Tire (RLT) model which is the dynamic tire model. First, the existing bicycle cornering kinematics that assumes the steady state is reformulated as a model considering the yaw acceleration, a transient characteristic. Second, the target yaw-rate considering the RLT model serves to compensate the phase delay. The proposed method can contributes to securing the performance and lateral stability of the Integrated Chassis Control(ICC) system by increasing the responsiveness of the model to the level of the actual vehicle.

After investigating the suitability of the vehicle motion simulation, it is also investigated the influence of the control input required by using the direct yaw moment control when applying it as the supervisor of the chassis control algorithm.

The proposed method has been investigated under several standard maneuvers via simulation with CarSim vehicle dynamics software and Matlab/Simulink and vehicle test data. The results show the proposed target yaw-rate which is incorporating transient handling characteristics well represents natural vehicle response such as phase delay and agility from mild handling maneuver to the limit handling maneuver. It has also been confirmed that it can alleviate the sense of difference that the driver felt from the existing over-control.

Keywords: Chassis control, Target yaw-rate design, Vehicle stability control, Lateral vehicle dynamics, Transient handling characteristics
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Chapter 1

Introduction

1.1 Background and Motivation

To improve the convenience of drivers, many chassis control systems such as Active Front Steering (AFS), Rear Wheel Steering (RWS), Electronic Control Suspension (ECS), and Active Roll Control System (ARS) are being assessed to determine how to achieve the greatest efficiency [Her15]. Frequently, at least two chassis control systems are applied to one vehicle to improve vehicle performance, and many studies have shown the integration of individual modular chassis control systems. In order to cope with the complicated operation conditions and to improve vehicle stability, maneuverability, ride comfort and/or vehicle safety, various chassis control systems have been developed and equipped in vehicles one after another.

The one of major goals of these systems include improve vehicle safety, maneuverability, especially in adverse driving situations. And these performance of the vehicle are important in limit handling maneuvers as well as mild handling maneuvers. To enhance the performance of extreme driving, many researches have been studied such as chassis control algorithm, driver model and optimal racing profile, etc [Lee06, Nagai02]. However, these studies have only studied the individual chassis systems or upper-level controls, there have been few cases of intensive research on supervisor of algorithms that monitor vehicle states and calculate the desired(reference) behavior. If the desired motion is not correct, no matter how good the control strategy is, it is useless. Therefore, it is important to study the target motion design that can be referenced in response to the natural vehicle motion.

1.2 Purpose of Research

Development of Chassis module last decades enhance from maneuverability to vehicle stability. In recent years, there have been many researches integrating individual chassis control modules. In order to design effective chassis control algorithm, numerous researches based on Model-based control method was conducted. In model based control, target yaw-rate determines the performance of chassis module and the stability of the vehicle.

One of the widely used methods for yaw-rate design is the delay transfer function (1^{st} or 2^{nd} order) between steering wheel angle (SWA) and yaw-rate [Smith15, Rajamani11, Jung14, Fetrati16]. The desired yaw-rate expression can be rearranged in the forms of SWA, vehicle dynamic parameters which is combined with the steady-state equation for yaw-rate and understeer gradient [Rajamani11]. The yaw-rate design is accomplished by tuning the understeer gradient through a linear fitting between the SWA and lateral acceleration in a constant circular maneuver. Although the above method has the merit of being efficient to a particular scenario, there is a phase delay that varies depending on the driving scenario. To cope with this problem and to have versatility, the time constant is used as an adaptive parameter to adjust the delay [Jung14].

In this work, the algorithm consists of a model which is considered transient cornering characteristic and Relaxation Length Tire (RLT) model. To avoid the uncertainty of tuning the time constant for each scenario, modified model that derives time constant from bicycle model is proposed. The bicycle model, however, not incorporating transient characteristic has phase delay in process from steering angle to yaw-rate. In this respect, Relaxation Length Tire (RLT) model is considered [Koo06]. The proposed algorithm of this study is validated via computer simulation using Carsim and Matlab/Simulink and vehicle test data.

Chapter 2

Analysis of Vehicle Dynamics

2.1 Lateral Vehicle Dynamics and Kinematics

The Lateral vehicle dynamics is described based-on the Bicycle model with steady-state [Abe09, Rajamani11].

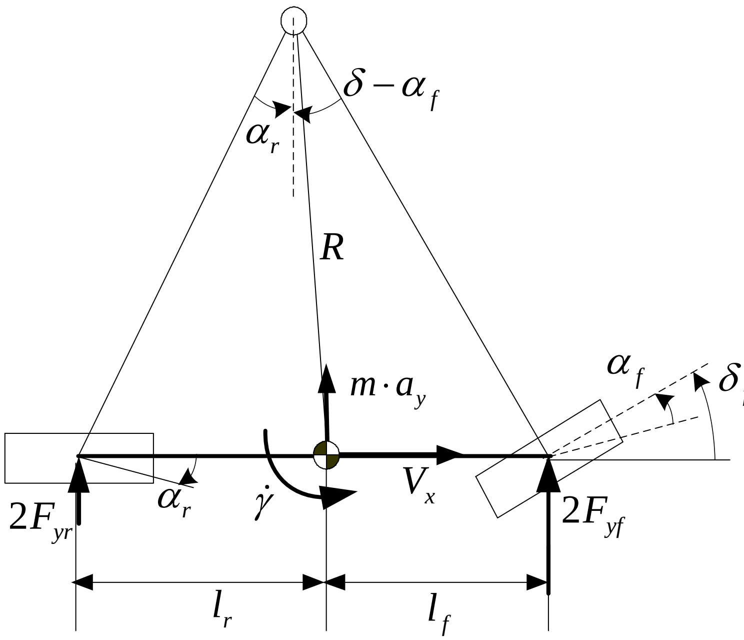


Figure 1. Bicycle model for cornering

In steady-state, the yaw-rate and lateral acceleration are expressed as follows:

$$\gamma = \dot{\psi} = \frac{V_x}{R} \tag{1}$$

$$a_y = \dot{V}_y + V_x \cdot \gamma \approx V_x \cdot \gamma = \frac{V_x^{\,2}}{R} \tag{2}$$

where ψ is the yaw-angle of the vehicle body, γ is the yaw-rate of the vehicle body, R is the radius of curvature, V_x is the vehicle longitudinal velocity, V_y is the vehicle lateral velocity, a_y is the vehicle lateral acceleration.

The steady-state front steering angle is given by kinematics as follow:

$$\delta_f \approx \frac{L}{R} + \alpha_f - \alpha_r \tag{3}$$

where δ_f is the steering angle of the front tire which is steering wheel angle(SWA) divided by gear ratio, L is the distance between the front axle and rear axle, $\alpha_f\,(\alpha_r)$ is the slip angle of the front (rear) tire.

The steady-state force and moment equilibrium equations are given by dynamics as follow:

$$\Sigma F_y = 2F_{yf} + 2F_{yr} = m \cdot a_y \tag{4}$$

$$\Sigma M_z = 2F_{yf} \cdot l_f - 2F_{yr} \cdot l_r \approx 0 \tag{5}$$

where F_y is the lateral force of the vehicle body, $F_{yf}\,(F_{yr})$ is the lateral force of the front(rear) tire, m is the total mass of the vehicle, M_z is the yaw moment of the vehicle, $l_f\,(l_r)$ is the distance between the center of gravity(C.G) and front (rear) axle.

The lateral tire forces are arranged through the alliance of (4) and (5):

$$\begin{aligned} F_{yf} &= \frac{ml_r}{2L} \cdot a_y \\ F_{yr} &= \frac{ml_f}{2L} \cdot a_y \end{aligned} \tag{6}$$

The lateral tire forces can be approximated in the linear region as follows:

$$\begin{aligned} F_{yf} &\approx C_f \cdot \alpha_f \\ F_{yr} &\approx C_r \cdot \alpha_r \end{aligned} \tag{7}$$

where C_f (C_r) is the cornering stiffness of the front (rear) tire in the linear region.

From (6) and (7), tire slip angles are described as follows:

$$\begin{aligned} \alpha_f &= \frac{ml_r}{2C_f L} \cdot a_y \\ \alpha_r &= \frac{ml_f}{2C_r L} \cdot a_y \end{aligned} \tag{8}$$

By substituting the above equations into (3), steady-state front steering angle is described as follows:

$$
\begin{aligned}
\delta_f &\approx \frac{L}{R} + \alpha_f - \alpha_r \\
&= \frac{L}{R} + \left(\frac{ml_r}{2C_f L} - \frac{ml_f}{2C_r L} \right) \cdot a_y = \frac{L}{R} + K_{us} \cdot a_y
\end{aligned} \tag{9}
$$

where K_{us} is the understeer gradient which is a parameter that allows to determine the steering sensitivity according to its sign.

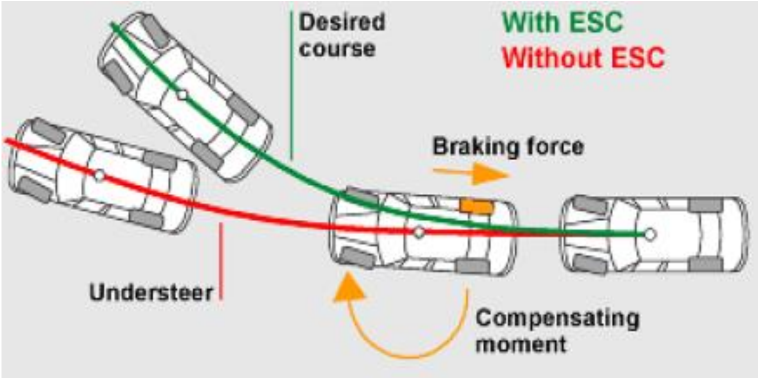
2.2 Vehicle Stability Control

It is important to enhance and secure the vehicle stability when the driver is in a dangerous situation with critical vehicle dynamics. Driver's load will be reduced if there is a control system that actively supports. As part of that, simple Electronic Stability Control(ESC) logic is used in this work [Shibahata94, Shino01, Yasui96].

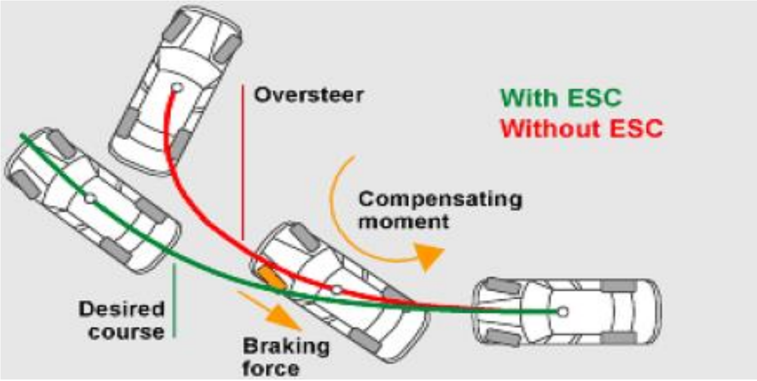
The objective of ESC is as follows:

- Through interventions in the braking system or the motor management
- To prevent critical situations, i.e., skidding, from leading to an accident
- To minimize the risk of side crash

It intervenes through identifying the driver's intention by driver-operator commands(position of the steering wheel, wheel speed, position of accelerator, and brake pressure) and perceive critical situations.



(a) Under Steer

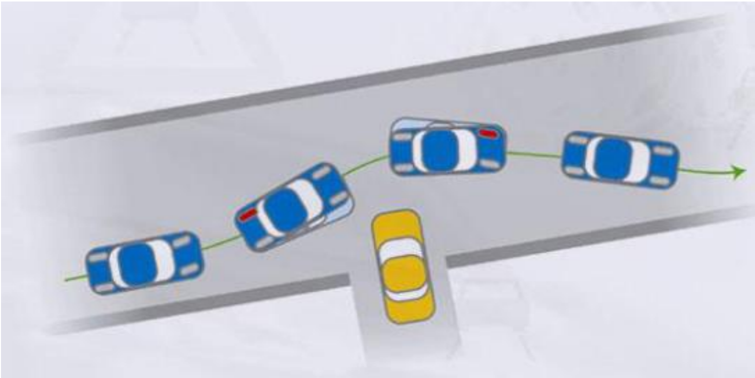


(b) Over Steer

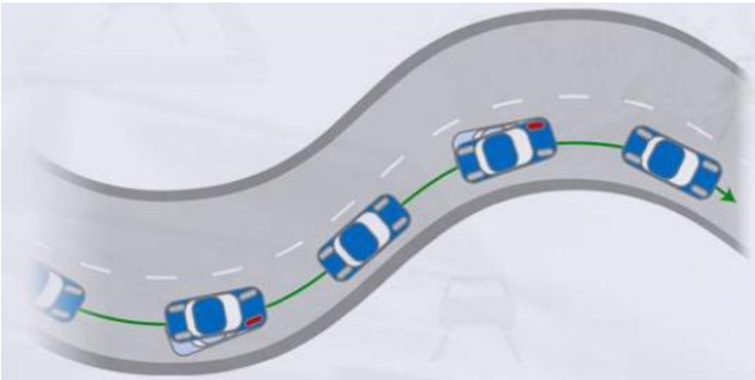
Figure 2. Effect of ESC according to steering status

A typical example of situation where ESC is required is as follows:

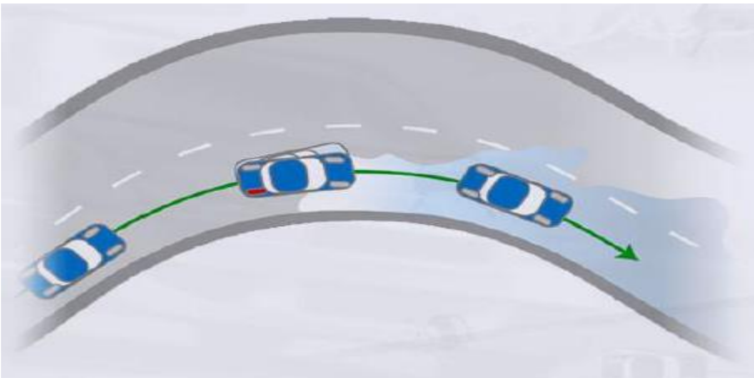
- Avoiding an obstacle
- Sudden wrenching of the steering wheel
- Driving on varying road surfaces(Longitudinal and/or Lateral changes)



(a) Avoiding an obstacle



(b) Sudden wrenching of the steering wheel



(c) Driving on varying road surfaces

Figure 3. Typical examples of situation where ESC is required

The conventional control algorithm for ensuring maneuverability of the vehicle is as follows [Cho11].

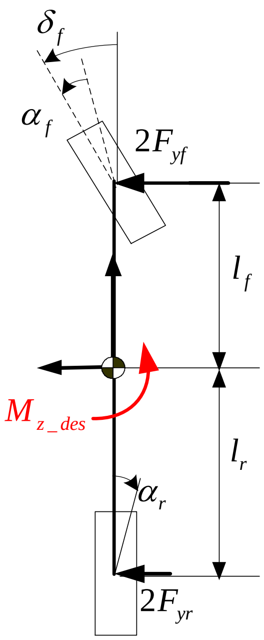


Figure 4. 2 DOF Bicycle model including Yaw moment input

$$x = \begin{bmatrix} \beta \\ \gamma \end{bmatrix} \quad \dot{x} = A \cdot x + B \cdot \delta_f + F \cdot M_{z\_des}$$

$$A = \begin{bmatrix} -\frac{2(C_f + C_r)}{m \cdot v_x} & -1 - \frac{2(l_f \cdot C_f - l_r \cdot C_r)}{m \cdot {v_x}^2} \\ -\frac{2(l_f \cdot C_f - l_r \cdot C_r)}{I_z} & -\frac{2({l_f}^2 \cdot C_f - {l_r}^2 \cdot C_r)}{I_z \cdot v_x} \end{bmatrix} \tag{10}$$

$$B = \begin{bmatrix} \frac{2C_f}{m \cdot v_x} \\ \frac{2l_f C_f}{I_z} \end{bmatrix} \qquad F = \begin{bmatrix} 0 \\ \frac{1}{I_z} \end{bmatrix}$$

Where M_{z_des} is the Yaw Moment control input based on Sliding Control Method [Yoshioka99, Zhao07, Zhu14].

Desired Yaw rate for Maneuverability.

$$\gamma_{desired} = \begin{cases} \gamma_{des,conventional}, & if \left|\gamma_{ss}\right| < \frac{\mu g}{v_x} \\ \frac{\mu g}{v_x} \cdot \mathrm{sgn}(\gamma_{ss}), & \text{elsewhere} \end{cases} \tag{11}$$

Define sliding surface.

$$s = (\gamma - \gamma_{desired}) \tag{12}$$

Lyapunov Function

$$V = \frac{1}{2}s^2 \tag{13}$$

Differentiating the Lyapunov function

$$\dot{V} = s \cdot \dot{s} \tag{14}$$

For stable system, the derivative of the Lyapunov function should be negative

$$\begin{aligned} &\dot{V} = s \cdot \dot{s} = -K \cdot |s| < 0 \\ &\therefore \dot{s} = -K \cdot \mathrm{sgn(s)} \end{aligned} \tag{15}$$

Substituting equation (10) and (12) into (14)

$$\begin{aligned} \dot{s} &= (\dot{\gamma} - \dot{\gamma}_{desired}) \\ &= -\dot{\gamma}_{desired} - \frac{2(l_f C_f - l_r C_r)}{I_z} \cdot \beta - \frac{2(l_f{}^2 C_f - l_r{}^2 C_r)}{I_z v_x} \cdot \gamma + \frac{2 l_f C_f}{I_z} \cdot \delta_f + \frac{M_{z\_des}}{I_z} \\ &= -K \cdot \mathrm{sgn(s)} \end{aligned} \tag{16}$$

Rewritten the above equation

$$M_{z\_des} = I_z \cdot \left(\frac{2(l_f C_f - l_r C_r)}{I_z} \cdot \beta + \frac{2(l_f{}^2 C_f - l_r{}^2 C_r)}{I_z v_x} \cdot \gamma - \frac{2 l_f C_f}{I_z} \cdot \delta_f \right) \quad (17)$$
$$- I_z \cdot K \cdot \mathrm{sgn(s)} + I_z \cdot \dot{\gamma}_{des}$$

The saturation function is used to cope with the chattering phenomenon as follows:

$$M_{z\_des} = M_{z\_\mathrm{eq}} - I_z \cdot K \cdot sat(\frac{\dot{\gamma} - \dot{\gamma}_{des}}{\Phi})$$
$$where,\ M_{z\_eq} = I_z \cdot \left(\frac{2(l_f C_f - l_r C_r)}{I_z} \cdot \beta + \frac{2(l_f{}^2 C_f - l_r{}^2 C_r)}{I_z v_x} \cdot \gamma - \frac{2 l_f C_f}{I_z} \cdot \delta_f \right)$$
(18)

The Yaw moment input obtained as above is converted to brake pressure to be used for feedforward control in the vehicle.

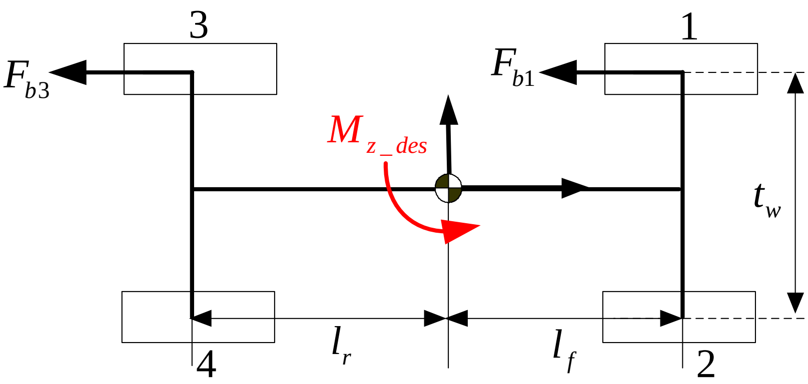


Figure 5. Planar model: Yaw moment to Brake pressure

The overall algorithm is as follows:

① If $M_{z\_des} > 0$

Only the 1, 3 wheels have brake pressure → $P_{b2} = P_{b4} = 0$

$$\begin{cases} M_{z\_des} = \dfrac{t_w}{2} \cdot \left(F_{b1} + F_{b3}\right) = \dfrac{t_w}{2R_{eff}} \cdot \left(T_{b1} + T_{b3}\right) \\ T_{b1} = p \cdot T_{b3}, \quad p : 0 \sim 1 \text{ portion} \end{cases}$$

Where R_{eff} is the effective radius, F_b is the braking force, T_b is the braking torque, P_b is the brake pressure.

$$T_{b1} = p \cdot T_{b3}$$

$$T_{b3} = \frac{2R_{eff}}{(1+p)t_w} M_{z\_des}$$

$$\therefore P_{b1} = \eta_{PT1} \cdot T_{b1}$$
$$P_{b3} = \eta_{PT3} \cdot T_{b3}$$

Where η_{PT} is the pressure to torque ratio.

② If $M_{z\_des} < 0$

Only the 2, 4 wheels have brake pressure → $P_{b1} = P_{b3} = 0$

Like ①,

$$T_{b2} = p \cdot T_{b4}$$
$$T_{b4} = \frac{2R_{eff}}{(1+p)t_w} M_{z\_des}$$
$$\therefore P_{b2} = \eta_{PT2} \cdot T_{b2}$$
$$P_{b4} = \eta_{PT4} \cdot T_{b4}$$

Chapter 3

Supervisor of Chassis Control System

In recent years, various studies have been carried out to improve the stability and maneuverability of the vehicle by equipped chassis control modules.

Inter alia, the chassis control modules associated with vehicle yaw-motion and their effects are briefly described as follows [Her15]:

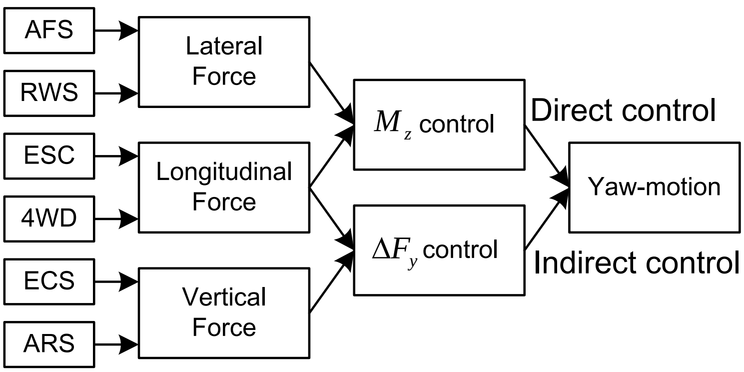


Figure 6. Chassis control modules' effects on yaw-motion

where AFS is Active Front Steering that generates direct yaw-moment by additional front steering angle, RWS is Rear Wheel Steering that generates direct yaw-moment by rear steering angle, ESC is Electronic Stability Control that generates direct yaw-moments by four independent brakes, 4WD is Four Wheel Drive that generates indirect yaw-moment by distributing traction between front and rear axle, ECS is Electronic Controlled Suspension that generates indirect yaw-moment by four controllable suspension, ARS is Active Roll Stabilizer that generates indirect yaw-moment by auxiliary roll moment.

The Integrated chassis control algorithm consisting of the modules is as follows:

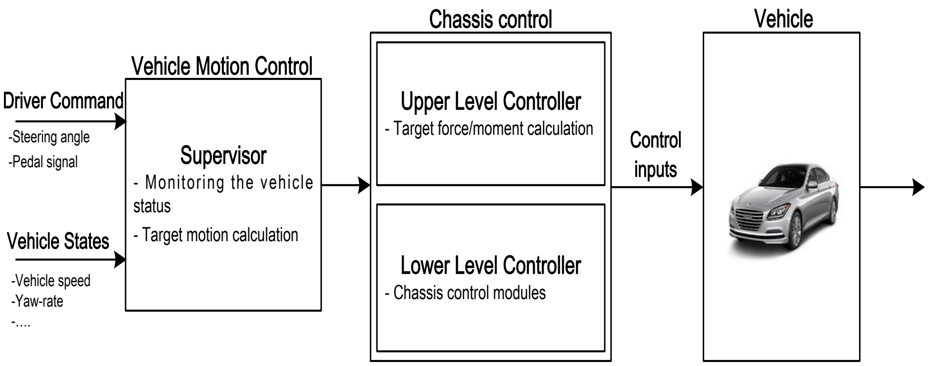


Figure 7. Integrated Chassis Control Algorithm

The algorithm consists of three parts: Supervisor, Upper Level Controller and Lower Level Controller. First, the supervisor monitors vehicle status and calculates the target yaw-rate based-on driver commands and sensor signals (vehicle states). Second, the upper level controller calculates the target force/moment to track the target motion. Finally, the lower level controller sends out the optimal control inputs based on upper level commands.

Like this, the target yaw-rate which is determined by supervisor is crucial value for chassis control performance. Therefore, the target yaw-rate must be calculated to ensure vehicle stability and maneuverability on various roads and maneuverings.

3.1 Conventional Target Yaw-rate Design

From steering kinematics in section 2, the steady-state yaw-rate is determined as follows:

$$\begin{aligned} \delta_f &= \frac{L}{R} + K_{us} \cdot a_y \\ &= \left(\frac{L}{V_x} + K_{us} \cdot V_x \right) \cdot \gamma_{ss} \end{aligned} \tag{19}$$

$$\begin{aligned} \gamma_{ss} &= \underbrace{\frac{V_x}{L + K_{us} \cdot V_x^{\,2}}}_{steady-state\ yaw-rate\ gain} \cdot \delta_f \\ &= G^{\gamma}{}_{ss} \cdot \delta_f \end{aligned} \tag{20}$$

where $G^{\gamma}{}_{ss}$ is the steady-state yaw-rate gain, K_{us} is the understeer gradient that is determined as the initial slope of SWA and lateral acceleration curve when constant circular turning scenario.

The target yaw-rate, which is commonly used, has the following relationship with the front steering angle input:

$$\gamma_{des} = G^{\gamma}{}_{ss} \cdot G^{\gamma}{}_{tr} \cdot \delta_f \tag{21}$$

$$G^{\gamma}{}_{tr} = \begin{cases} \dfrac{1}{\tau \cdot s + 1} & ,1^{st}\ order\ delay \\ \dfrac{1 + \tau_{\gamma} \cdot s}{1 + \dfrac{2\zeta}{\omega_n} \cdot s + \dfrac{1}{\omega_n{}^2} \cdot s^2} & ,2^{nd}\ order\ delay \end{cases} \tag{22}$$

where $G^{\gamma}{}_{tr}$ is the transient characteristics yaw-rate gain that is commonly 1st order [Smith15, Rajamani11, Jung14] or 2nd order delay [Matsumoto92, Fetrati16] form in transfer function, τ is the delay parameter, ζ is the damping ratio, ω_n is the natural frequency.

The transient characteristics yaw-rate gain is needed because the vehicle dynamics have non-linearity characteristics and actuators have delays. This is a design parameter that is determined by optimization in some criteria which ensure the agility of the vehicle. This parameter optimization is efficient in simulating the motion of a particular vehicle in a particular scenario, but it must be re-optimized if the road and maneuvering change. In other words, it has no generality for various scenarios.

3.2 Modified Target Yaw-rate Design

In this work, the target yaw-rate is re-calculated by considering the transient handling characteristics to reduce the uncertainty of the design parameter and to have versatility.

3.2.1 Transient handling characteristics

The force and moment equilibrium equations considering transient handling characteristics are given as follows:

$$\Sigma F_y = 2F_{yf} + 2F_{yr} = m \cdot a_y \tag{23}$$

$$\Sigma M_z = 2F_{yf} \cdot l_f - 2F_{yr} \cdot l_r = I_z \cdot \dot{\gamma} \tag{24}$$

where I_z is the moment of inertia on the yaw axis, $\dot{\gamma}$ is the yaw-acceleration of the vehicle.

The lateral tire forces are arranged through the alliance of (23) and (24):

$$\begin{aligned} F_{yf} &= \frac{ml_r}{2L} \cdot a_y + \frac{I_z}{2L} \cdot \dot{\gamma} \\ F_{yr} &= \frac{ml_f}{2L} \cdot a_y - \frac{I_z}{2L} \cdot \dot{\gamma} \end{aligned} \tag{25}$$

By substituting (7), (25) into (3), front steering angle is described as follows:

$$\begin{aligned} \delta_f &\approx \frac{L}{R} + \alpha_f - \alpha_r \\ &= \frac{L}{R} + \left[\underbrace{\left(\frac{ml_r}{2C_f L} - \frac{ml_f}{2C_r L} \right)}_{steady-state} + \underbrace{\left(\frac{1}{2C_f} + \frac{1}{2C_r} \right) \cdot \frac{I_z \cdot \dot{\gamma}}{L \cdot a_y}}_{transient\ handling\ characteristic} \right] \cdot a_y \\ &= \frac{L}{R} + \left(K_{us\_ss} + \Delta K_{us} \right) \cdot a_y \end{aligned} \tag{26}$$

where K_{us_ss} is the steady-state understeer gradient, ΔK_{us} is the additional term of understeer gradient by yaw-acceleration (transient handling characteristic).

3.2.2 Dynamic Tire Model

Since the tire forces do not develop instantaneously, lag is needed into the slip angle such that the instantaneous response calculated for the lagged slip angle yields the lagged lateral force [Bernard95, Koo06, Loeb90].

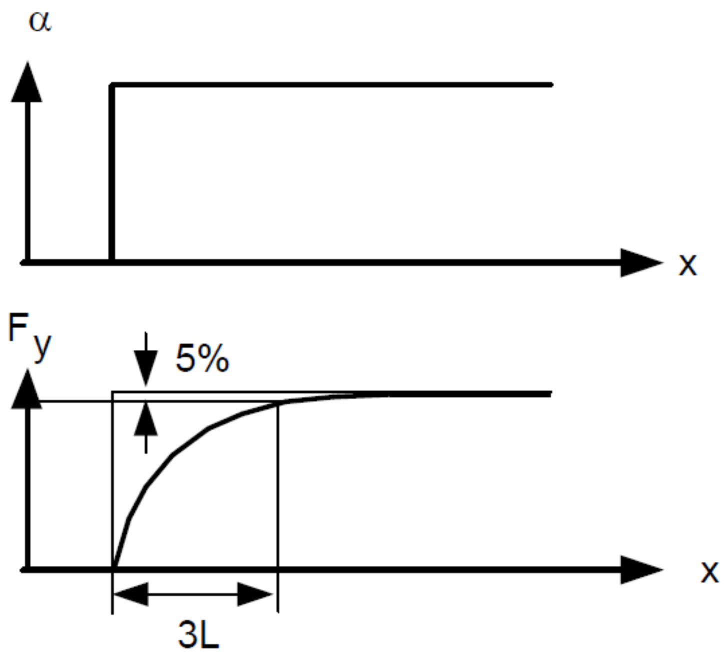


Figure 8. Relaxation Length Tire (RLT) model

The lagged slip angle is defined as follows:

$$\frac{d\alpha_{lag}}{dt} = \frac{V_x}{L_y} \cdot \left(\frac{V_y}{V_x} - \alpha_{lag} \right) \tag{27}$$

where L_y is the relaxation length for lateral slip.

With the assumption that the slip angle is small enough, the effect of RLT can be summarized in s- domain (Laplace transform) as follows:

$$s\alpha_{lag}(\mathrm{s}) = \frac{V_x}{L_y} \cdot \left(\underbrace{\frac{V_y}{V_x}}_{\approx\alpha} - \alpha_{lag}(\mathrm{s}) \right) \tag{28}$$

$$\begin{aligned} &\left(\underbrace{\frac{L_y}{V_x}}_{\tau} s + 1 \right) \cdot \alpha_{lag}(\mathrm{s}) = \alpha(\mathrm{s}) \\ &\therefore \alpha_{lag}(\mathrm{s}) = \frac{1}{\tau s + 1} \cdot \alpha(\mathrm{s}) \end{aligned} \tag{29}$$

where τ is the ratio of relaxation length and longitudinal velocity that functions like a time constant.

The effect of RLT on lateral tire forces is as follows:

$$
\begin{aligned}
F_{yf} &= C_f \cdot \alpha_f \cdot \frac{1}{\tau s + 1} \\
F_{yr} &= C_r \cdot \alpha_r \cdot \frac{1}{\tau s + 1}
\end{aligned}
\tag{30}
$$

The process of obtaining the target yaw-rate by RLT and transient handling characteristic is as follows:

$$
\begin{aligned}
& s - domain \\
& \delta_f = \frac{L}{R} + \alpha_f - \alpha_r = \frac{L}{R} + \left(\frac{F_{yf}}{C_f} - \frac{F_{yr}}{C_r} \right) \cdot (\tau s + 1) \\
& = \frac{L}{V_x} \cdot \Gamma + \left[K_{us\_ss} \cdot V_x \cdot \Gamma + \left(\frac{1}{C_f} + \frac{1}{C_r} \right) \cdot \frac{I_z}{2L} \cdot s\Gamma \right] \cdot (\tau s + 1) \\
& = \left(\frac{L}{V_x} + K_{us\_ss} \cdot V_x \right) \cdot \Gamma + \left[K_{us\_ss} \cdot V_x \cdot \tau + \left(\frac{1}{C_f} + \frac{1}{C_r} \right) \cdot \frac{I_z}{2L} \right] \cdot s\Gamma \\
& + \left(\frac{1}{C_f} + \frac{1}{C_r} \right) \cdot \frac{I_z}{2L} \cdot \tau \cdot s^2 \Gamma
\end{aligned}
\tag{31}
$$

where Γ is the yaw-rate on s-domain.

This is converted back from the Laplace domain to the time domain as follows:

$$t-domain$$

$$\delta_f = \left(\frac{L}{V_x} + K_{us\_ss} \cdot V_x\right) \cdot \gamma \tag{32}$$

$$+\left[K_{us\_ss} \cdot V_x \cdot \tau + \left(\frac{1}{C_f} + \frac{1}{C_r}\right) \cdot \frac{I_z}{2L}\right] \cdot \dot{\gamma} + \left(\frac{1}{C_f} + \frac{1}{C_r}\right) \cdot \frac{I_z}{2L} \cdot \tau \cdot \ddot{\gamma}$$

$$\text{Define } state\ X = \begin{bmatrix} \gamma \\ \dot{\gamma} \end{bmatrix} \qquad input\ u = \delta_f \tag{33}$$

$$\dot{X} = A \cdot X + B \cdot u$$

$$A = \begin{bmatrix} 0 & 1 \\ -\frac{2C_f C_r L}{(C_f + C_r) I_z \tau} \cdot \left(\frac{L}{V_x} + K_{us\_ss} \cdot V_x\right) & -\frac{2C_f C_r L \cdot K_{us\_ss} \cdot V_x}{(C_f + C_r) I_z} - \frac{1}{\tau} \end{bmatrix} \tag{34}$$

$$B = \begin{bmatrix} 0 \\ \frac{2C_f C_r L}{(C_f + C_r) I_z \tau} \end{bmatrix}$$

Target yaw-rate

$$\gamma_{des} = \begin{bmatrix} 1 & 0 \end{bmatrix} \cdot X \tag{35}$$

Through the above process, total ICC algorithm flow applied to the supervisor is as follows:

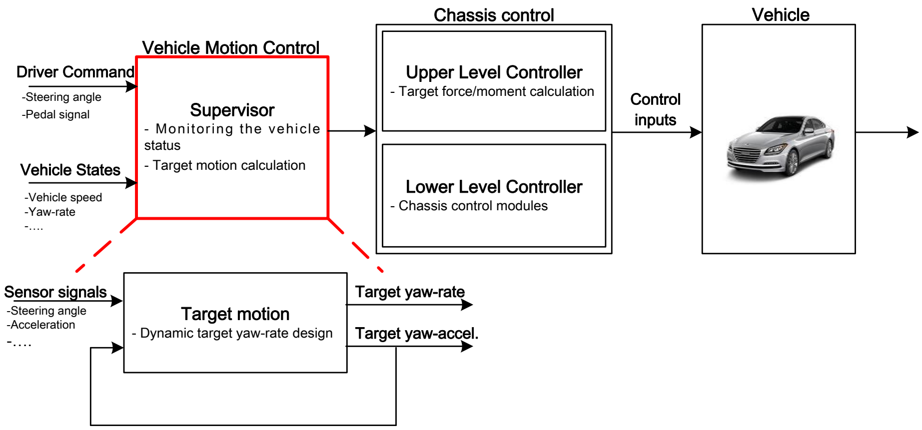


Figure 9. Overall scheme of ICC algorithm

Chapter 4

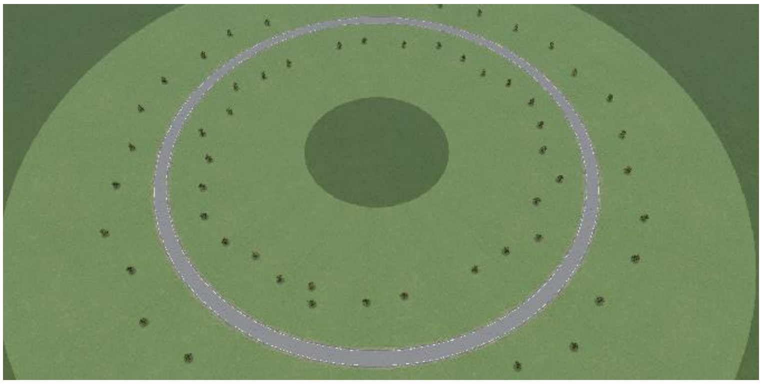
Comparison / Validation

To validate generality of the proposed target yaw-rate, simulation and vehicle test have been conducted by open-loop driver-vehicle system subject to constant circular turning with acceleration, slalom test which represents mild handling maneuver and lane change which represents limit handling maneuver. Simulations using CarSim the vehicle dynamics software and Matlab/Simulink.

The proposed algorithm is validated via simulation and vehicle test data under the following standard scenarios:

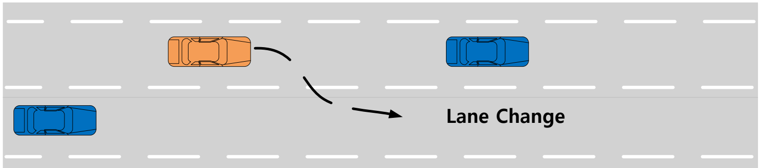
Table 1. Simulation/Vehicle test Scenarios

Scenario	Speed	Lateral acceleration (peak)
Constant Circular Turning with Acceleration (R=45m)	0 → 65kph	0 → 0.8g
Slalom (Mild maneuver)	75kph	0.1g
Lane change (Limit handling maneuver)	75kph	0.8g



(a) Scenario 1 – Constant Circular Turning (R=45m) with acceleration

(b) Scenario 2 – Slalom test (Mild handling maneuver)



(c) Scenario 3 – Lane change (Limit handling maneuver)

Figure 10. Test scenarios

Since the vehicle test has been conducted with the Genesis G80, simulation also has been performed using the relevant parameters in the following table.

Table 2. Vehicle parameters based on G80

Notation	Parameter	Value
m	Vehicle mass	2273 kg
L	Wheel base	3.010 m
l_f	Distance from C.G to front axle	1.456 m
I_z	Moment of inertia on z-axis	5112 $kg \cdot m^2$
C_f	Front cornering stiffness	1885 $N/\deg$
C_r	Rear cornering stiffness	3675 $N/\deg$
L_y	Relaxation length for lateral slip	0.565 m

4.1 Validation of Target yaw-rate design

4.1.1 Scenario 1: Constant Circular Turning with Acceleration

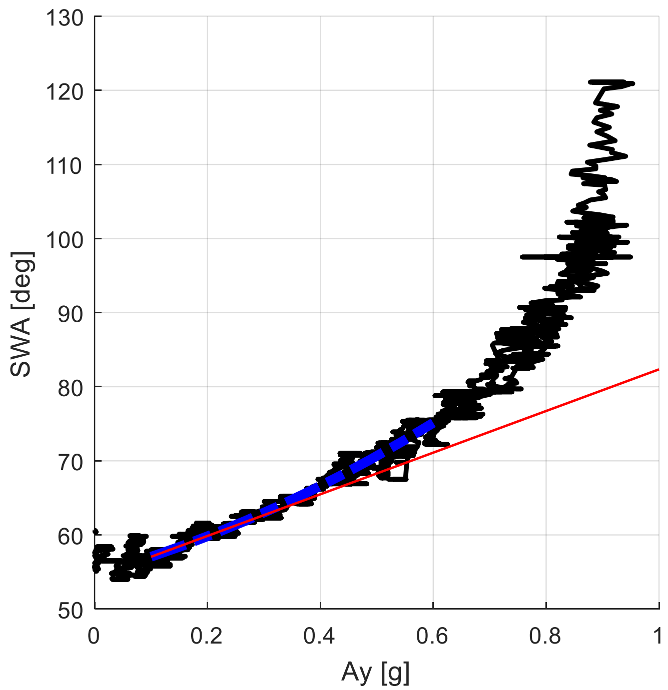


Figure 11. SWA vs. Ay

Constant circular turning with acceleration scenario is essential to find the steady-state understeer gradient of vehicle.

This is the definition of steady-state understeer gradient(K_{us_ss}) as follows:

$$\begin{aligned}\delta_f &\approx \frac{L}{R}+\alpha_f-\alpha_r \\ &= \frac{L}{R}+\left(\frac{ml_r}{2C_fL}-\frac{ml_f}{2C_rL}\right)\cdot a_y = \frac{L}{R}+K_{us\_ss}\cdot a_y\end{aligned} \tag{36}$$

From this, the slope of Figure 7 means steady- state understeer gradient. It is necessary to tune the steady-state gain of conventional target design. In this work, it is fitted the linear region ($A_y \le 0.4g$) as shown by the red line in the Figure 7. This means that the conventional target yaw-rate, which is to be compared with proposed method, is designed to correspond to the mild handling maneuver.

4.1.2 Scenario 2: Mild Handling Maneuver

To verify whether the proposed method has compatibility between different scenarios, simulations and vehicle tests have been conducted for normal driving and extreme driving. The analysis is performed by comparing the results of the conventional method and the proposed method based on the behavior of the basic vehicle with no chassis control.

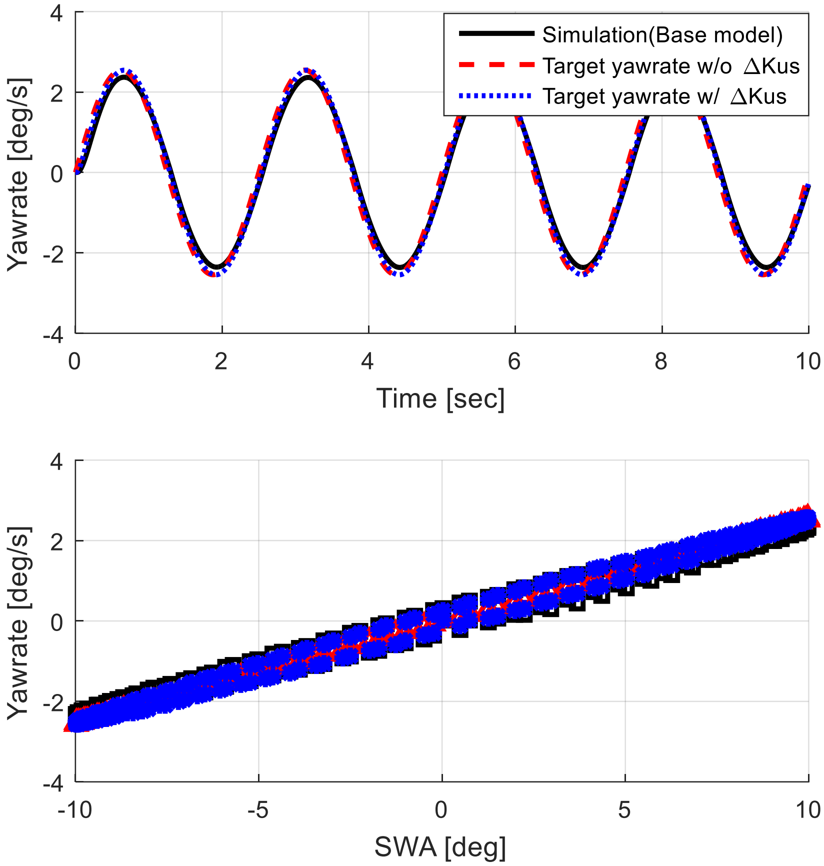


Figure 12. Simulation results: Vehicle response with Mild maneuver

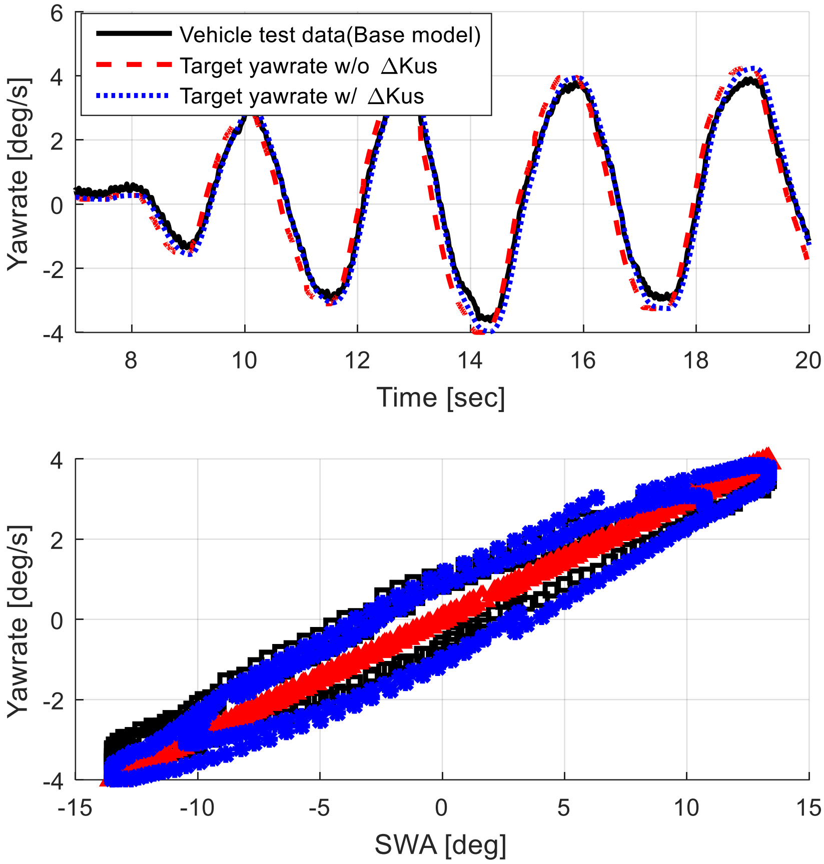


Figure 13. Vehicle test results: Vehicle response with Mild maneuver

Where the base model means no chassis control vehicle, w/o ΔK_{us} means conventional design method, w/ ΔK_{us} means proposed design method.

In the figure, each graph shows as follow:

- Top graph of figure: Comparing the degree of yaw-rate simulation for each design method
- Bottom graph of figure: Vehicle agility which indicates the vehicle's response to the driver input.

It seems that there is no significant difference in yaw-rate response, either in vehicle test or simulation. In the case of agility, simulation result looks the same as above, but the proposed method seems to be more complementary in the vehicle test result.

4.1.3 Scenario 3: Limit Handling Maneuver

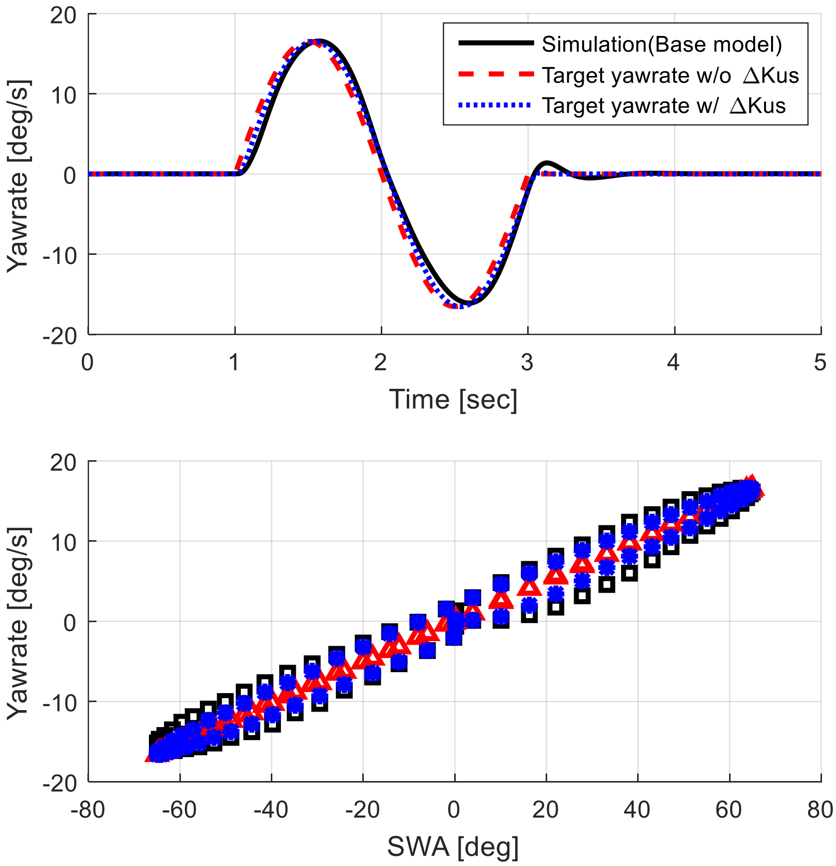


Figure 14. Simulation results: Vehicle response with Limit handling maneuver

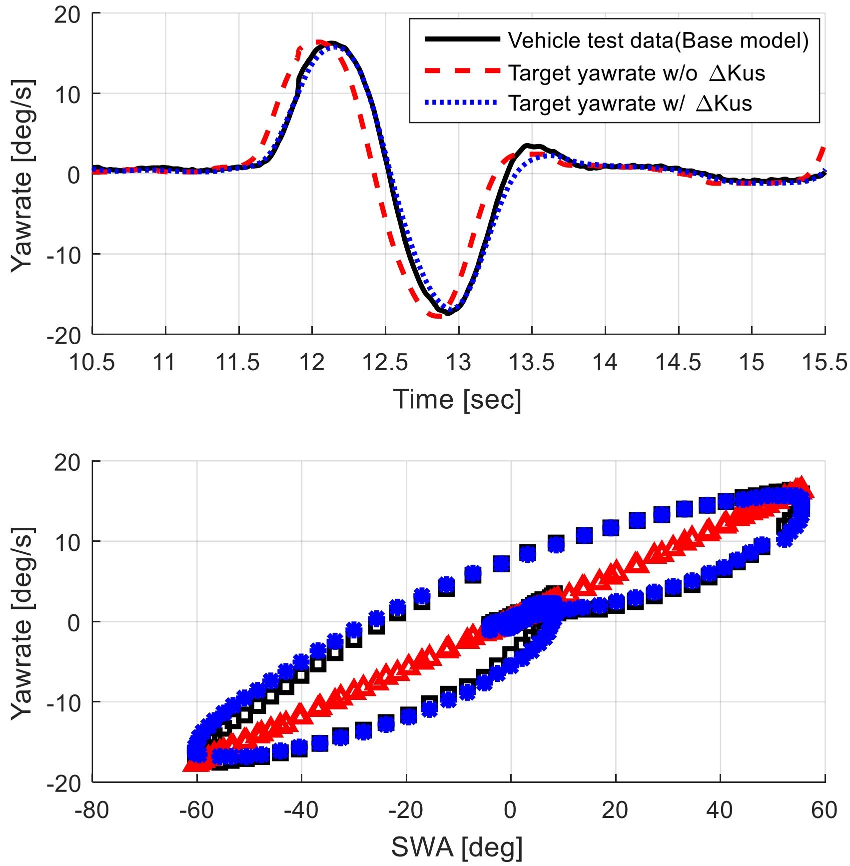


Figure 15.Vehicle test results: Vehicle response with Limit handling maneuver

Limit handling maneuver means driving with high lateral acceleration level. As a result, conventional target do not fit well because lateral acceleration of scenario goes beyond the designed range. On the other hand, it seems that the proposed target method is improved both in terms of yaw-rate response and vehicle agility. Especially, it is noticeably improved in vehicle test result.

4.2 Performance of Target yaw-rate design

To verify the effect of target yaw-rate accuracy on the chassis control module, conventional target yaw-rate and proposed target yaw-rate are applied to the ESC mentioned in section 2.2 as an example.

The comparison of the yaw moment input from the Lane change scenario which is the limit handling maneuver is as follows:

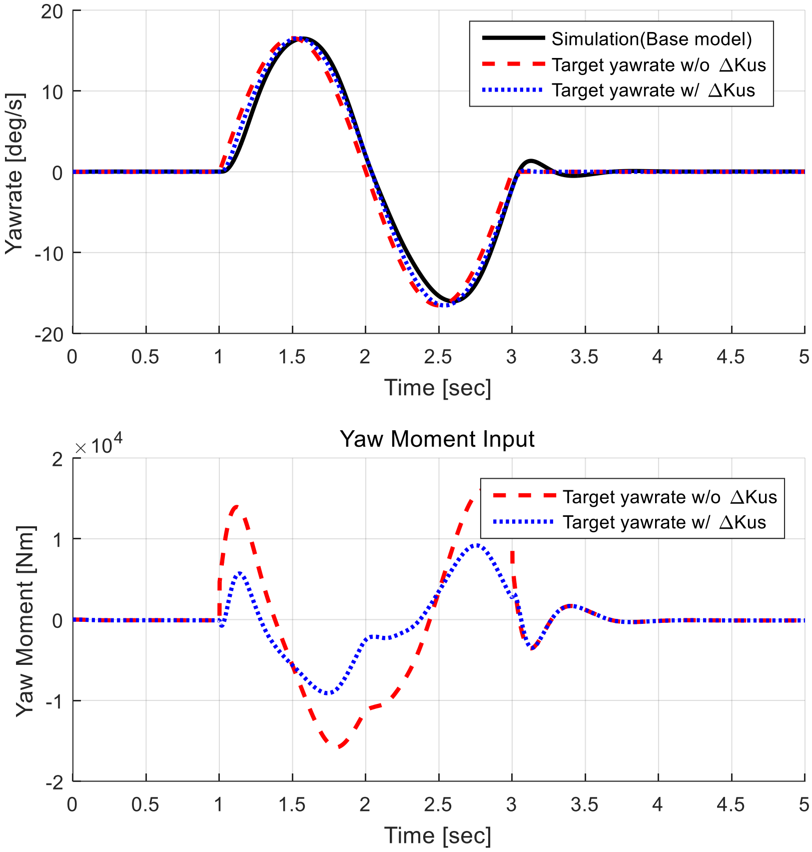


Figure 16. Simulation results: Control input with Limit handling maneuver

Excessive control input has been given to the conventional target design because it was not possible to simulate the actual vehicle motion in extreme driving. In this respect, the proposed target design seems to have mitigated that degree.

Chapter 5

Conclusion and Future Work

A dynamic target yaw-rate design for chassis control, which is considering the transient handling characteristics, is presented in this thesis. Numerous researches have been conducted to improve the stability and maneuverability of the vehicle equipped with the chassis control module, and the target yaw-rate design that determines the performance is important. Rather than using a design parameter to compensate the delay, the target yaw-rate is determined by vehicle dynamics considering the transient characteristic and dynamic tire model. The proposed target design method has been investigated under mild/limit handling scenarios via simulation and vehicle test. The target is validated by comparing vehicle responses consists of yaw-rate response and vehicle agility. In slalom test, there is no significant difference between the two methods because the conventional target are designed for the mild handling area. In limit handling maneuver with lateral acceleration which is more than 0.4g, the conventional target does not fit well but the proposed target is improved. Additionally, it can be seen that the vehicle test results are more noticeable than simulation results. Therefore, it can be concluded that the proposed target yaw-rate design has generality for various scenarios compared to the conventional target yaw-rate design.

Then, a simple chassis control (ESC in this work) has been performed and the control input according to each method has been compared. It seems that the proposed method could solve the sense of difference which is from too frequent strong control input when using the conventional method.

In addition to ESC above, the proposed target yaw-rate should be investigated how it affects the performance of various actuators such as AWD, RWS, ECS when it plays a role as the supervisor of chassis control system.

In this study, the purpose of the target yaw-rate design is to have versatility in various scenarios. It is assumed that there are several handling situations for the scenarios, but all of them proceeded in the high friction surfaces. In the future, additional research should be done to see if the content shown in this work can be applied in low friction surfaces.

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초 록

샤시 제어를 위한 동적 목표 요레이트 설계

본 논문은 샤시 제어 시스템을 위한 동적 목표 요레이트 설계 방법을 제시한다. 목표 요레이트는 통합 샤시 제어(ICC) 알고리즘의 슈퍼바이저에 있어 필수적이다. 슈퍼바이저는 차량 상태를 모니터링하고 목표 요레이트와 같이 목표하는 차량 거동을 결정한다. 횡력, 요모멘트와 같은 제어입력이 상위 및 하위 레벨의 컨트롤러에서 목표 거동에 따라 계산되므로 목표 설계가 중요하다. 기존의 목표 설계는 특정 시나리오 및 도로 조건에 대한 파라미터 최적화를 통해 이루어졌다. 그러나 이것은 서로 다른 시나리오 사이에 호환성이 결여되어 있다는 단점이 있다.

이 논문에서는 목표 요레이트가 범용성을 포함하도록 만드는 연구가 진행되었다. 제안된 설계 방법은 핸들링에서의 과도특성을 포함하는 자전거 모델과 동적 타이어 모델인 Relaxation Length Tire (RLT) 모델로 이루어진다. 첫째, 정상 상태를 가정한 기존의 코너링 동역학은 과도 특성인 요가속도를 고려한 모델로 재구성된다. 둘째, RLT 모델을 고려함으로써 목표 요레이트는 위상 지연을 보완하게 된다. 제안된 설계 방법은 모델의 응답성을 실제 차량 수준으로 끌어올려 통합 샤시 제어 시스템의 성능과 횡방향 안정성을 확보하는데 기여할 수 있다. 차량 거동 모사의 적합성을 조사한 후,

Direct yaw moment 제어 알고리즘의 슈퍼바이저에 적용했을 때 제어입력에 어떠한 영향을 주는지를 조사한다.

제안된 방법은 표준 시나리오를 CarSim 차량 동역학 소프트웨어 및 Matlab/Simulink를 이용한 시뮬레이션과 실차 시험을 통해 검증이 진행되었다. 결과는 핸들링에서의 과도 특성을 포함하고 있는 제안된 목표 요레이트가 가벼운 핸들링 조작에서부터 극한 주행에 해당되는 핸들링 조작까지 위상 지연 및 차량의 민첩성과 같은 측면에서 차량 거동을 잘 모사할 수 있음을 확인할 수 있다. 또한, 운전자가 기존의 과도한 제어로부터 느꼈던 이질감을 완화할 수 있었음을 확인했다.

주요어: 샤시 제어, 목표 요레이트 설계, 횡방향 차량 동역학, 선회시 과도 특성, 차량 안정성 제어

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